Effects of Contact Charging on Spray Impingement Heat Transfer Performance and Spray Behavior

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Abstract

The effects of contact charging on the heat transfer performance of a full-cone spray of the dielectric coolant, HFE-7000, has been studied using a Thick Film Resistor (TFR) heater with an active surface area of 1.46 cm² at a nozzle-to-heater spacing of 13 mm. Tests have been conducted at coolant flow rates between 1.3 and 6.8 GPH (1.4 x 10⁻⁶ to 7.1 x 10⁻⁶ m³/s), for heater power ranging from 0 to 60 Watts, yielding heat fluxes between 0 and 400 kW/m². Voltage levels applied to the brass spray nozzle to charge the spray range from 0 to 30 kV, using both negative and positive polarity.

A dramatic change in the visual spray flow pattern is observed as the charging voltage exceeds approximately 15 kV in magnitude. Above this voltage, the spray changes from droplets and collections of clearly observable discrete sheets to what appears to primarily be a finer mist of smaller droplets between the spray nozzle and the heater surface. This is due to exceeding the Rayleigh limit for the maximum charge on the liquid droplets, resulting in electrostatic atomization to a smaller average droplet size. However, no significant change in the measured heat transfer performance is seen between the no voltage and the high voltage cases up to a maximum charging voltage of 30 kV.

Nomenclature

\[ b = \text{radius of the heater (m)} \]
\[ G\Delta = (\text{Heater Power}) / [\pi b (T_{\text{sat}} - T_{\infty,\text{wall}}) k_{\text{tr}}], \text{ non-dimensional heater power} \]
\[ h = \text{heat transfer coefficient (W/m}^2\text{°C)} \]
\[ k_{\text{fluid}} = \text{thermal conductivity of the working fluid (W/m°C)} \]
\[ k_{\text{tr}} = \text{thermal conductivity of the heater (W/m°C)} \]
\[ \text{Nu} = \text{Nusselt number, defined as } h b / k_{\text{fluid}} \]
\[ T_s = \text{heater surface temperature (°C)} \]
\[ T_{\text{sat}} = \text{liquid saturation temperature (°C)} \]
\[ T_{\infty,\text{top}} = \text{average temperature of liquid on heater surface (°C)} \]
\[ T_{\infty,\text{wall}} = \text{average temperature of the liquid in sump (°C)} \]
\[ \Theta_s = (T_s - T_{\infty,\text{wall}})/(T_{\text{sat}} - T_{\infty,\text{wall}}) = \text{non-dimensional heater surface temperature} \]
\[ \Theta_{\infty}\text{-top} = (T_{\infty,\text{top}} - T_{\infty,\text{wall}})/(T_{\text{sat}} - T_{\infty,\text{wall}}) = \text{non-dimensional liquid film temperature on surface of heater} \]

I. Introduction

There is a constant push to achieve higher heat transfer rates, both for commercial applications (eg., supercomputer cooling, or PC CPU cooling) and military applications (eg., aircraft and spacecraft component cooling). Both Beam (2000) and Mahefkey et al. (2004) discuss possible future military aircraft cooling requirements that would require heat rejection of on the order of 100 kW-1 MW of waste
Heat. Spray cooling is one of several potential methods that are being developed for high heat flux applications, where Mudawar (2000) identifies spray cooling as having the potential for developing the highest heat flux for a given coolant of the various methods reviewed.

Spray cooling significantly increases the cooling spatial uniformity relative to liquid jet impingement cooling (Bernardin et al., 1996). Spray cooling has the best performance in terms of heat transfer coefficient and critical heat flux (CHF) when compared to liquid jet impingement and pool boiling (Tilton, 1989; Chow, Sehmbey, and Pais, 1997; Mudawar, 2000). With water as the working fluid, spray cooling has achieved a heat flux on the order of 1000 W/cm² in terrestrial gravity (Lin and Ponnappan, 2003).

The application of electrical body forces, or Electro-Hydro-Dynamics (EHD) to enhance fluid heat transfer has been under study for some time; see the reviews by Jones (1978) and Seyed-Yagoobi and Bryan (1999). The EHD effects on the flow and heat transfer are due to the creation of body forces that are spatially nonuniform and can locally be quite large. Castellanos (1991) discusses the application of EHD Coulomb forces to drive convective liquid flows. Significant EHD heat transfer enhancements have been observed in both pool boiling and in flow boiling.

However, to date very little work has been done on the application of EHD forces to enhance spray cooling heat transfer rates or CHF. Feng and Bryan (2005) are studying the basic physics of EHD liquid jet and spray impingement. Baysinger et al. (2004) and Yerkes et al. (2006) at the Air Force Research Laboratory (AFRL) describe a spray impingement heat transfer experiment that is being flown on NASA variable gravity research aircraft, in an effort to achieve higher heat transfer performance in a weight and space efficient manner. The coolant used in their experiments is FC-72, a nonpolar, dielectric, diamagnetic liquid. The present work is a continuation of studies conducted at West Virginia University over the past three years to explore the effectiveness of the electrical body forces to enhance spray impingement heat transfer in the laboratory (Gray et al., 2007). For this work, a spray cooling experiment has been developed that is identical to the AFRL apparatus, but with provision for generating controlled electric fields near the spray nozzle and heater surface. Heat transfer and flow visualization results are reported herein for contact charging of the spray to create a Coulomb EHD force that attracts the spray to the heater surface. A schematic of the spray nozzle and heater surface, indicating the direction of the resulting Coulomb force on a charged spray droplet is shown in Figure 1. Note that the direction of this force is towards the heater surface regardless of the spray nozzle polarity.

![Figure 1](image-url)
II. Previous Work

Successful application of EHD forces to enhance heat flux has been reported by several authors; see the surveys by Jones (1978) and Seyed-Yagoobi and Bryan (1999). One early example of the use of EHD forces to enhance boiling heat transfer is the patent by Chubb (1916). Enhancements have been reported both for pool boiling (Ogata et al., 1990 and 1992; Ohadi et al., 1992; Damianidis et al., 1995; Seyed-Yagoobi et al., 1996; Karayiannis, 1998; Chen and Liu, 1999; Huang et al., 2004; Yajima et al., 2004) and for flow boiling (Wang et al., 1996; Salehi et al., 1997; Bryan and Seyed-Yagoobi, 2000; Cotton et al., 2002 and 03; Grassi et al., 2005). Using several different refrigerants or dielectric coolants, enhancements in pool boiling of four- to eight-fold increases in heat flux have been observed by these authors at lower heat fluxes (5-15 kW/m²). However, this EHD enhancement typically is reduced significantly at higher heat fluxes. EHD enhancement factors reported for flow boiling are typically much lower than for pool boiling, with reported enhancement values from 10% to 100%, but these heat flux levels are much higher than for pool boiling. One exception is a reported ten-fold EHD enhancement in small 1 mm grooved channels at a Reynolds number of 300 (Salehi et al., 1997). Darabi et al. (2000) have studied EHD enhancement of falling-film evaporation on horizontal tubes. Snyder et al. (2001) have used applied voltages of 6-16 kV to obtain bubble detachment for pool boiling in microgravity using two different electrode geometries. DiMarco and Grassi (2002) have demonstrated significant increases in the CHF in pool boiling of FC-72 via the use of the electric Kelvin force at electric potentials of 10 kV, both in Earth gravity and microgravity. Yang et al. (2002) used EHD forces to enhance heat pipe performance, and Darabi (2004) describes the EHD enhancement of microscale thin film evaporation.

Chow et al. (1997) and Kim (2006) have summarized spray cooling research. Chen et al. (2002) have studied the effects of altering the droplet velocity, droplet diameter, and droplet number flux on heat flux and CHF for more than 20 different full-cone nozzle designs. Varying the three parameters for each nozzle yielded more than 3000 different combinations. Varying the droplet diameter while holding number flux and velocity constant did not show a clear trend for either CHF or heat transfer coefficient. Increasing the number flux from $7 \times 10^6$ to $29 \times 10^6$ (cm$^2$/s)$^{-1}$ increased the CHF about 30%, while the heat transfer coefficient increased about 20%. Droplet number flux herein is $90 \times 10^6$ (cm$^2$/s)$^{-1}$ at the highest flow rate of 10 GPH (10.5 x 10$^{-6}$ m$^3$/s). Increasing the droplet velocity from 4.6 to 21.4 m/s increased CHF from 640 to 950 W/cm², an increase of nearly 50%. The heat transfer coefficient increased nearly 40%, from 6 to 8.4 W/cm²K. Thus, changing the droplet velocity has the greatest impact on the resulting CHF and heat transfer coefficient, with number flux also having a significant influence. Silk et al. (2007) have used these results as a guide to develop a physics-based correlation for spray cooling CHF.

Initial heat transfer results for the present spray cooling apparatus, without any EHD forces, have been reported by Hunnell (2005) and Hunnell et al. (2006). The 3M Corporation dielectric coolant, FC-72 was used in these studies. An initial study of the use of the electric Kelvin force has been summarized in the thesis by Glaspell (2006) and by Kreitzer et al. (2006). The use of the Coulomb force via inductive charging electrode designs (Law, 1978) is described in the thesis by Kreitzer (2006) and by Kuhlman et al. (2007). Both the Kelvin force studies by Glaspell (2006) and the inductive charging Coulomb force work by Kreitzer (2006) used FC-72 and a second 3M dielectric coolant, HFE-7000. Also, both of these studies demonstrated small (order of 5-15%) but repeatable EHD enhancements in heat flux using HFE-7000. EHD enhancements to heat flux were observed only during boiling conditions. These enhancements were also only observed for electrode designs that created localized EHD body forces that tended to repel the liquid from the heater surface. This was an unanticipated result, since body forces that attract the liquid to the heater surface, thereby repelling the vapor bubbles, have generally been used in both pool boiling and flow boiling studies. Lee et al. (1997) observed heat flux enhancements under conditions of reduced gravity in their pool boiling experiments.

III. Apparatus and Procedure

The spray nozzles, heaters, and sump geometry in the present work are identical to the spray cooling apparatus developed at the AFRL (Baysinger et al., 2004). Instead of using the coolant used in the AFRL work (FC-72; see Yerkes et al., 2006), a second dielectric coolant, HFE-7000, has been used due to its lower resistivity and higher dielectric constant (Gray et al., 2007). Detailed descriptions of the current apparatus have been given in the theses by Hunnell (2005), Glaspell (2006), and Kreitzer (2006). A schematic of the spray nozzle, heater surface, pedestal, and sump is shown in Fig. 2, along with a
photograph of the nozzle and heater. The nozzle and heater surface have been housed in a spray chamber that has been fitted with view ports for flow visualization. The geometry of this portion of the present apparatus is identical to the AFRL apparatus, except for electrical penetrations in the WVU spray chamber for the present study of the effects of Coulomb body forces on spray cooling performance.

For the Spraying Systems full cone 1/8G-1 brass nozzle used herein, Yerkes et al. (2006) have measured Sauter mean diameter and velocity at an FC-72 flow rate of 9.5x10^-6 m^3/s using Phase Doppler Anemometry as 48 µm and 12 m/s, respectively. Nozzle-heater spacing for the present results is 13 mm, and pedestal diameter is 16 mm. The nozzle-heater spacing was chosen so that the outer edge of the spray cone coincides with the circumference of the heater (Fig. 2). The heater active surface area is 1.46 cm^2.

Two types of ceramic Thick-Film Resistor (TFR) heaters have been used in the present work: one has a 40 µm thick glass layer bonded to the top of the heater (the TFR heaters used at AFRL), while the second custom version does not have this glass layer. The TFR heater without the glass layer has been used for the present heat transfer results. The heaters are bonded to the top of a cylindrical pedestal (Fig. 2); the pedestal is installed in a sump that is used to collect the excess liquid coolant for recirculation. The sump may be fitted with a concentric hollow conical cap to redirect the excess liquid into the sump and to aid in liquid flow management; however the present results have been obtained without a cap fitted to the sump. High voltage is supplied to the spray nozzle/electrode by a Glassman model EL30R1.5 reversible-polarity 0-30 kV power supply. Negative polarity has been used for the heat transfer results presented herein; similar heat transfer and spray behavior was observed using positive polarity.

The spray chamber has been mounted to a base that houses the necessary pumps, valves, heat exchangers, and instrumentation. A schematic of the apparatus flow loops housed in the experiment base is shown in Fig. 3. Coolant is pumped from a reservoir by a positive-displacement gear pump to the spray nozzle, where flow rate is controlled by setting the pump speed, and a rotameter monitors flow rate. The spray impinges on the heater surface, and the excess liquid is collected in the sump. The vapor is condensed on the chamber walls and is then also collected in the sump. A positive-displacement diaphragm pump sends the sump fluid through a liquid-air heat exchanger back into the reservoir. Flow returned to the sump is controlled by a valve on the bypass loop (Fig. 3). Spray chamber temperature is controlled by water flowing through tubes cemented to its outside cylindrical surface. This flow is created by a separate flow loop, consisting of a water reservoir, liquid-air heat exchanger, centrifugal pump, and rotameter. Both flow loops contain filters, as well as pressure and temperature instrumentation.

Seven 0.25 mm (0.01") type E thermocouples have been installed in the glass post onto which the heater has been mounted in the same locations as those used for the AFRL apparatus (Baysinger, 2004). A similar PTFE pedestal fitted with 5 thermocouples located directly beneath the TFR heater has also been used in the present work. The analysis developed by Yerkes et al. (2006) has been used to compute the
heater surface temperature from the measured temperature at a location 0.5 mm below the heater surface in contact with the coolant. Key thermocouples have been calibrated against an RTD standard thermometer in a stirred oil bath. Flow meter repeatability is estimated as ±3% to 5%, heater power accuracy is estimated as ±0.5 to 1 W, and temperature resolution is ± 0.1 °C, with an estimated accuracy of ±0.2 °C for all calibrated thermocouples and ±0.6 °C for uncalibrated thermocouples (Kreitzer, 2006).

Dimensional performance results have been analyzed in the non-dimensional form developed by Yerkes et al. (2006). The heat flux has been non-dimensionalized as the parameter, $G \Delta$ (Yerkes et al., 2006), defined as:

$$G \Delta = \frac{\text{Heater Power}}{\pi b (T_{\text{sat}} - T_{\infty,\text{wall}}) k_{\text{ht}}}. \quad (1)$$

Here, $b$ is the radius of the heater, $T_{\infty,\text{wall}}$ is the average temperature of the spray liquid as it travels through the sump, and $k_{\text{ht}}$ is the thermal conductivity of the heater material. The temperature difference, $(T_s - T_{\infty,\text{top}})$, has been non-dimensionalized by dividing by $(T_{\text{sat}} - T_{\infty,\text{wall}})$. The reference temperature $T_{\infty,\text{top}}$ is the average temperature of the liquid film on the heater surface, taken as the average of the initial spray liquid temperature and the temperature of the liquid as it leaves the heater surface. $T_s$ is the heater surface temperature, computed from the measured interface temperature at the bottom of the heater substrate. The heat transfer model developed by Yerkes et al. (2006) has been extended in an approximate way to include the effect of the insulating glass layer on the top side of the TFR ceramic heater that is fitted with this glass in the calculation of the heater surface temperature, $T_s$ (Hunnell et al., 2006). A simple one-dimensional conduction heat transfer thermal resistance across the insulating glass layer on the side of the TFR heater in contact with the spray coolant has been used to compute the temperature drop across this glass layer in order to find $T_s$. This results in a reduction of the heater surface temperature across the TFR heater with the glass layer of approximately 5 °C at a heater power of 60 W. The computed heat transfer coefficients have been non-dimensionalized as the Nusselt number, Nu, defined as:

$$\text{Nu} = \frac{h b}{k_{\text{fluid}}}. \quad (2)$$

Here, $k_{\text{fluid}}$ is the fluid thermal conductivity and $h$ is the heat transfer coefficient.
IV. Results

Spray cooling performance results for the present apparatus without electrical body force effects have been presented by Hunnell (2005) and by Hunnell et al. (2006) under conditions of terrestrial gravity for both vertically-downward directed and horizontal sprays. Effects of the Coulomb force via inductive charging electrode designs (Law, 1978) have been described in the thesis by Kreitzer (2006), while effects of electrodes using the electric Kelvin force have been presented in the thesis by Glaspell (2006). The present results using the Coulomb force via contact charging are consistent with these earlier results. First, a series of high-speed video visualizations of the interaction between the impinging spray and liquid film on the heater surface will be described, followed by a comparison of the heat transfer results with and without contact charging of the spray.

In Fig. 4, two example frames are shown from high-speed digital video observations at 25,000 frames per second using laser light sheet visualization techniques and a Phantom v4.2 digital high-speed camera at 10 µs exposure to yield a “frozen” image of the spray droplet pattern, indicating that the droplets become significantly smaller at the higher charging voltages. For this comparison the heater power was zero for both videos, spray flow rate was 4.1 GPH (4.3 x 10^{-6} m^3/s), and the laser light sheet was oriented vertically to cut through the centerline of the spray. Playback of the video files at reduced speed (160x) indicates clearly the swirling of the spray, with the largest spray droplets moving in an expanding spiraling pattern until impacting the heater surface. Also, this spiral pattern appears to consist primarily of ligaments and/or sheets of liquid rather than individual droplets, and these ligaments/sheets appear to be significantly larger than the 48 µm droplet diameter measured using PDA (Yerkes et al., 2006). It is believed that existence of these large ligaments or sheets of liquid may result in inaccuracies in the PDA droplet diameter measurements of Yerkes et al. (2006). As a result, the time scale analysis by Kuhlman et al. (2007) should be revised in light of the presence of the large features in the spray. Similar video files recorded at 9,600 frames per second with better spatial resolution that also show the heater surface indicate significant splashing of these larger droplets, ligaments, and sheets when they impact the heater. This splashed liquid moves radially outwards at a velocity that is from 5 to 10 times slower than the nominal 10 m/s droplet velocity; some of the splashed liquid moves upwards away from the heater (“rebound”), some moves radially parallel to the surface, and some is influenced by surface tension to turn downwards at the edge of the heater to flow down the pedestal into the sump.

For 48 µm HFE-7000 liquid droplets, the computed Rayleigh charging limit (Law, 1978) is approximately 6 x 10^6 e/drop. The observed current drain at an electrode voltage of 30 kV and the droplet Sauter mean diameter measured without the electric field would also yield a charge of 6 x 10^6 e/drop at this flow rate. This is consistent with the conclusion that, at the higher charging voltages, the observed current flow must exceed the Rayleigh charge limit, and many of the droplets become electrostatically atomized to...
diameters still smaller than the 48 µm mean diameter reported by Yerkes et al. (2006) for the uncharged spray.

Figs. 5, 6, and 7 present comparisons of the behavior of the liquid film on the heater surface with and without heating and with and without the effects of the high voltage at an HFE-7000 flow rate of 2.3 GPH (2.4 x 10^{-6} m^3/s). Frame rate was 2,200 frames per second, exposure time was 444 µs, and arc lamp lighting from the opposite side of the chamber was used. At this relatively long exposure setting, it is not possible to freeze the spray motion, so the spray droplets appear as long streaks in the images. The liquid film on the heater surface is observed to be extremely irregular and contorted for all three cases, both due to the impinging spray, and due to boiling for Figs. 6 and 7. Slow-motion analysis of video shows the liquid film moving radially at nominally 0.5-1 m/s.

Figure 5. Four Successive Frames from High-Speed Video Visualization at 2,200 Frames per Second of HFE-7000 Spray at Flow Rate of 2.3 GPH, Zero Heater Power and 0 kV.

Figure 6. Four Successive Frames from High-Speed Video Visualization at 2,200 Frames per Second of HFE-7000 Spray at Flow Rate of 2.3 GPH, T_1 = 63 °C and 0 kV.

Figure 7. Four Successive Frames from High-Speed Video Visualization at 2,200 Frames per Second of HFE-7000 Spray at Flow Rate of 2.3 GPH, T_1 = 63 °C and 20 kV.
Heating at 45 W yields large transient patches of what appear to be frothy vapor bubbles (Figs. 6 and 7); these bubble patches also move in the radial direction. It appears that these bubbles primarily are caused to move radially outwards along the heater surface when large sheets/ligaments/droplets of the spray liquid impact the heater surface. Such impacts splash both the bubble patches and much of the impinging liquid outwards towards the outer edge of the heater. Again, some of the liquid film on the heater surface is able to turn the corner and move downwards into the sump along the cylindrical surface of the pedestal. These splashing events appear like breaking waves of foamy vapor bubbles.

Application of the 20 kV high voltage (Fig. 7) yields what appears to be a “fog” of smaller spray droplets, but the larger spiral patterns of ligaments or sheets of liquid that are seen without the high voltage applied are still visible in the spray pattern. These smaller droplets visible as a fog are formed by the electrostatic atomization of droplets once they reach the Rayleigh charging limit. Also, patches of vapor bubbles are still visible on the heater surface, similar to the no-voltage case at the same heater power. With the high voltage applied to the spray nozzle, there also is a continuous streaming of small droplets back from the heater surface towards the nozzle that is visible outside of the spray cone in the videos in which the laser light sheet cuts through the spray (Fig. 4, and also part (B.) of Fig. 8 at a lower flow rate). This is likely due to liquid that has impinged onto the heater surface acquiring positive charge as it flows over the heater prior to being splashed off of the surface in the form of small droplets. Note that at the lower spray flow rate of Fig. 8, while the fog of smaller droplets due to the Coulomb force is still clearly visible, the remnants of the ligaments and/or sheets of liquid are also still quite prevalent.

Cossali et al. (1997) have developed a criterion for the onset of splashing due to droplet impingement onto a surface covered with a preexisting liquid film. This criterion predicts no splashing for 48 µm Sauter mean diameter droplets, but somewhat larger droplets are predicted to splash. Cole et al. (2005) and Gray et al. (2007) have used computational fluid dynamics to simulate these phenomena for a single droplet impinging into a thin liquid film; their simulations agree with the experimental observations by Cossali et al. Clearly, for the present spray significant splashing is observed in the high-speed video observations.

Pautsch et al. (2004) have reported spray impingement cooling liquid film thicknesses ranging between 80 µm and 300 µm. It is estimated that the transient liquid film thickness for the present work lies somewhere in this range, away from regions where individual droplets have just impacted the heater surface. Much thinner film thicknesses (on the order of 1 µm) are estimated in these droplet impact regions (Kuhlman et al., 2007).

The electric potential around the nozzle electrode and the grounded heater surface (Fig. 2) has been simulated using the multiphysics code, CFD-ACE+; sample results at 10 kV are shown in Fig. 9. Similar simulations of the earlier inductive charging electrodes (Kuhlman et al., 2007) and Kelvin force electrodes

(A.) No heat transfer; 0 kV.

(B.) No heat transfer; 20 kV.

Figure 8. Two Sample Frames from High-Speed Video Visualization at 9,600 fps of HFE-7000 Spray at Flow Rate of 2.4 GPH, Zero Heater Power.
(Kreitzer et al., 2006) have been shown in the dissertation by Mehra (2007). From the present simulations, the electric field strength very near to the spray centerline has been computed, as plotted in Fig. 10 for a nozzle voltage of 10 kV. These results are reasonably close to the average electric field strength that is computed (700 kN/C at 10 kV) from the electrode charging voltage and the nozzle-to-heater spacing. These electric field results have been used to compute an average Coulomb force on a droplet having the 48 µm average diameter reported by Yerkes et al. (2006), using the Rayleigh limit charge of 6 x 10^6 e/drop (9.6 x 10^{-13} C/drop) to yield a force of 0.67 µN. Using the droplet volume of 5.8 x 10^{-14} m^3 yields a force per unit mass of 8,200 N/kg, a very significant force; this force is directed towards the heater surface.

![Electric Potential for Nozzle Voltage of 20 kV](image)

Figure 9. Electric Potential for Nozzle Voltage of 20 kV, Computed Using CFD-ACE+.

However, even though these computed Coulomb forces are quite significant, no significant change in the heat transfer performance has been observed for applied voltages between 0 to 25 kV, as may be seen in Figs. 11-16. In Fig. 11 the nondimensional heat flux, GΔ, is shown versus the nondimensional temperature difference between the heater surface and the impinging liquid spray for an HFE-7000 flow rate of 6.8 GPH (7.1 x 10^{-6} m^3/s). All five curves are virtually indistinguishable, indicating clearly no effect of the Coulomb force on the heat transfer, as well as good data repeatability. The corresponding Nusselt number results are shown versus GΔ in Fig. 12. Again, the results for all five voltage values are essentially identical.

Similar results at a spray flow rate of 4.2 GPH (4.4 x 10^{-6} m^3/s) shown in Figs. 13 and 14 again show no significant variations in the measured heat transfer performance as the contact charging electrode voltage is varied from 0 kV to 25 kV. Comparison between Figs. 11 and 13 indicate that the heater surface is hotter at the lower flow rate. Nusselt numbers are also reduced as flow rate is reduced (Fig. 12 and Fig. 14). The measured heat transfer data at a spray flow rate of 1.3 GPH (1.4 x 10^{-6} m^3/s) are presented in
Figs. 15 and 16. Again, no consistent trend in performance is seen as the contact charging voltage is varied. The heater surface runs still hotter (Fig. 15) and the heat transfer coefficient is reduced further (Fig. 16). An unusual kink in the heat flux-temperature difference behavior is seen in Fig. 15 at this lowest flow rate. The cause of this behavior is not known for certain, but it may be due to a gradual onset of CHF above a GΔ value of approximately 200 at this flow rate.

Figure 10. Electric Field Strength at Spray Centerline at Nozzle Voltage of 10 kV Computed from CFD-ACE+.

Given the dramatically different overall appearance of the spray when the high voltage is applied to the nozzle (Fig. 4, Fig. 7, and Fig. 8), and the large computed magnitude of the Coulomb force, it is quite surprising that there is no significant effect due to contact charging on the heat transfer performance for the present results. However, it is noted that the largest features visible in the spray (called ligaments and/or sheets above) are still visible in the video images even with the high voltage applied; see Fig. 4, Fig. 7, and Fig. 8. Also, it is the impact of these largest features of the spray with the liquid film on the heater surface that appears in the video to dominate the breakup and splashing off of the surface of the vapor bubble patches that are visible on the heater at high heat fluxes. It is therefore speculated that the impact of these large sheets and/or ligaments of spray liquid may be most significant in determining the heat transfer for this spray nozzle. Thus, it is expected that the length and time scales for the impacts of these large liquid ligaments or sheets with the heater surface should significantly influence the heat flux and especially CHF (Kuhlman et al., 2007). Also, for both the previous electric Kelvin force electrode results (Glaspell, 2006) and the previous inductive charging Coulomb force results (Kreitzer, 2006), the heat flux enhancements that were observed occurred only when the EHD forces on the droplets were directed away from the heater surface, while for the present contact charging Coulomb force results, the EHD force is directed towards the heater. Thus, future work should focus on developing electrode geometries that will lead to EHD forces that are directed away from the heater surface.

V. Conclusions

Contact charging effects on heat transfer performance of a full-cone spray of HFE-7000 has been studied using a heater with an active surface area of 1.46 cm$^2$ at a nozzle-to-heater spacing of 13 mm. Contact charging of the spray results in Coulomb forces on the order of several thousand N/kg that attract the spray towards the heater surface.

Tests have been conducted at coolant flow rates between 1.3 and 6.8 GPH (1.4 x 10$^{-6}$ to 7.1 x 10$^{-6}$ m$^3$/s), for heater power ranging from 0 to 60 Watts, yielding heat fluxes between 0 and 400 kW/m$^2$. Voltage levels applied to the brass spray nozzle range from 0 to 30 kV, using both negative and positive polarity.
Figure 11. Non-Dimensional Heat Flux Versus Temperature Difference Using Coulomb Force and TFR Heater for HFE-7000 at Flow Rate of 6.8 GPH.

Figure 12. Nusselt Number Versus Non-Dimensional Heat Flux Using Coulomb Force and TFR Heater for HFE-7000 at Flow Rate of 6.8 GPH.
Figure 13. Non-Dimensional Heat Flux Versus Temperature Difference Using Coulomb Force and TFR Heater for HFE-7000 at Flow Rate of 4.2 GPH.

Figure 14. Nusselt Number Versus Non-Dimensional Heat Flux Using Coulomb Force and TFR Heater for HFE-7000 at Flow Rate of 4.2 GPH.
Figure 15. Non-Dimensional Heat Flux Versus Temperature Difference Using Coulomb Force and TFR Heater for HFE-7000 at Flow Rate of 1.3 GPH.

Figure 16. Nusselt Number Versus Non-Dimensional Heat Flux Using Coulomb Force and TFR Heater for HFE-7000 at Flow Rate of 1.3 GPH.
The spray flow pattern changes significantly above charging voltages of approximately 15 kV. In addition to collections of clearly observable discrete sheets and droplets, a finer mist of smaller droplets is observed in the spray between the spray nozzle and the heater surface above these voltages. This is due to exceeding the Rayleigh limit for the maximum charge on the liquid droplets, resulting in electrostatic atomization to a smaller average droplet size. However, no significant change in the measured heat transfer performance is seen between the no voltage case and the high voltage cases up to a maximum voltage of 30 kV. This is consistent with earlier studies using the electric Kelvin force, where modest heat transfer enhancements were observed; for these earlier results, the EHD forces were directed away from the heater surface. Since the magnitude of the Coulomb force that can be achieved is significantly larger than that of the electric Kelvin force, future studies should concentrate on developing optimal methods of inductive charging of the spray. Further high-speed video imaging should be helpful in studying the physical mechanisms for both inductive charging and/or the electric Kelvin force to generate spray cooling heat flux enhancements when these forces are directed away from the heater surface.

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